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by Thomas N. Strom, Lawrence P. Ludwig,
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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ABSTRACT

In inert and reducing environments both at room and elevated temperatures, undamped seals operated with severe vibration which caused early bellows fatigue and high leakage rates of a 1.57-inch (3.99-cm) mean diameter seal operating to speeds of 165 feet per second (50.3 m/sec) at 24 000 rpm. Friction damping by mechanical dampers and the viscous damping by oil in the bellows were both effective in maintaining seal stability. Nosepiece temperatures caused by the sliding contact were high enough to reduce hardness of the mating seal seats.

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SUMMARY

Shaft face seals of 1.57 inch (3.99 cm) mean diameter with a bellows secondary seal were operated in inert and reducing gaseous environments. Starting temperatures ranged from 75° to 800° F (297 to 700 K) and sliding speeds ranged to 165 feet per second (50.3 m/sec) at 24 000 rpm. A study was made of seal stability (nosepiece vibration) as affected by both friction and viscous damping. The studies showed that undamped bellows seals have a very strong tendency for the nosepiece to vibrate in an unstable manner at high frequency (400 Hz) and high amplitudes (0.010 in. or 0.025 cm). This severe vibration allows excessive leakage because of the large nosepiece amplitudes and can cause early bellows failure. In contrast, seals with either friction (mechanical dampers) or viscous damping operated in a stable manner from 75° to 800° F (297 to 700 K) and to sliding speeds of 165 feet per second (50.3 m/sec) at 24 000 rpm. Stable seals, when operating without lubrication, however, had high (>700° F or 644 K) nosepiece temperatures; lubricated seals had significantly lower nosepiece temperatures. Results of allied studies showed that the temperature generated by the sliding contact can be high enough to reduce seat metal hardness.

INTRODUCTION

Small shaft face seals of 1.57 inches (3.99 cm) mean sliding diameter with a bellows secondary seal are being used in small gas turbine engines, rocket turbopumps, and turbosuperchargers. These seals operate at high shaft speeds (to 60 000 rpm) in a hot (to 900° F or 755 K) gaseous environment. In the case of gas turbine engines, the hot gases are oxidizing; in rocket turbopump drives the gases are reducing; and in turbosuperchargers the gases are either oxidizing or reducing, depending on load. In these gaseous environments, an undamped bellows seal with one of its natural frequencies

near shaft rotational frequency could operate in an unstable manner; that is, the seal nosepiece (stator) could vibrate and cause a marked increase in seal leakage (ref. 1). More significantly, reference 2 reports severe nosepiece vibrations, for seals operating in liquid nitrogen, at frequencies not in resonance with shaft speed. In the study of reference 2, the vibration amplitude of the nosepiece was measured to be as large as 0.037 inch (0.094 cm) with predominant frequencies of 500 to 600 hertz. With such large amplitudes, the leakage past the nosepiece sealing dam is excessive; the high frequencies, coupled with the large amplitudes, quickly induce bellows fatigue failures.

The source or forcing function which induces the nosepiece vibration could be shaft axial vibration (ref. 3), seat face runout (ref. 3), or frictional forces at the seal sliding interface. Of these three sources, the data of reference 2 suggests that the severe vibrations are more readily caused by frictional forces at the sliding interface. Thus the coefficient of friction associated with sliding material combinations and the environment may affect seal vibration. For instance, it has been recognized (ref. 4) that carbon-graphites have lower friction coefficients in oxidizing environments than in inert or reducing environments. Thus, larger frictional forces may exist in inert or reducing environments.

The objectives of this study were to (1) determine if a shaft face seal (undamped) operates unstably (nosepiece vibration) in inert and reducing environments, (2) determine the effect of friction (coulomb) and viscous damping on seal vibration and leakage rate, and (3) measure seal temperature induced by high-speed operations.

Shaft face seals to 1.57 inches (3.99 cm) mean diameter at the sliding interface were operated to speeds of 165 feet per second (50.3 m/sec) at 24 000 rpm in inert and reducing environments. Both friction and viscous damping were employed to inhibit vibration. The sealed pressure differentials were generally small in the range of 2 pounds per square inch (1.38 N/cm^2). Vibration amplitude, frequency, leakage rate, and temperature were measured.

APPARATUS AND PROCEDURE

Typical Application

Figure 1 is a schematic drawing of a typical application of a shaft face seal in a rocket engine turbopump. The function of the face seal is to restrict the leakage of hot products of combustion into the area containing the oil lubricated support bearings. The bellows secondary seal acts to hold the nosepiece in sliding contact against the rotating seat and permits axial displacements due to thermal expansion.

Room Temperature Apparatus, Nitrogen Gas Environment

For seal vibration studies at room temperature the application depicted in figure 1 was simulated by the apparatus of figure 2. A nitrogen gas environment was used. The shaft which carries the rotating seat is the stub end of a high-speed grinding spindle. The test seal has an insert cemented to the nosepiece inside diameter, and a displacement probe monitors the axial movement (vibration) of this insert. As a second means of sensing vibration, subminiature accelerometers were also cemented to this insert. A circumferential shaft riding seal was used to form a cavity which was pressurized with nitrogen gas to a gage pressure of 2 pounds per square inch (1.38 N/cm^2). Leakage through the seal was measured by a rotating drum wet test meter. Temperature of the carbon-graphite seal nosepiece was measured by means of thermocouples embedded in the carbon-graphite. The thermocouple wire size was small, 0.003 inch (0.0076 cm), in order not to interfere with seal dynamics. No external heating was used; thus any increase in the carbon-graphite temperature was due to sliding at the interface. In some runs, Mil-L-7808D lubricant was introduced through a feed tube at the rate of 1 cubic centimeter per minute.

Elevated Temperature Apparatus

In tests at ambient temperatures to 800° F (700 K), the apparatus sketched in figure 3 was employed. The rotating seat was mounted on a gas bearing supported spindle and face runout was held within 0.0005 inch (0.0018 cm) full indicator runout (FIR). The nosepiece and bellows assembly carried by the housing could be positioned axially, thus changing the amount of bellows compression. The seal assembly and gas environment was heated by means of an induction coil surrounding the seal support housing. Leakage through the seal was collected in the enclosure and measured by passing the leakage through a wet test meter. Mil-L-7808D lubricant could be supplied by means of a small tube (fig. 3) directing lubricant flow to the seal interface. Shaft speed was monitored by a magnetic pickup and counter. One of the gases used in the elevated temperature apparatus was termed products of combustion (POC) which had the following analysis: 24.7 percent CH_4 , 0.7 percent CO, 0.7 percent CO_2 , 40.0 percent H_2 , 25.7 percent N_2 , and 8.2 percent H_2O .

Face Seal With Bellows Secondary

Figure 4 consists of schematics of the shaft face seals used in sealing nitrogen gas and products of combustion. The nosepiece was carbon-graphite impregnated with silver, and the seat was 17-4 PH stainless steel. A 300 series stainless steel was used for the seal housing, nosepiece carrier, and bellows. To establish the seal operating length, the bellows spring rate was measured, and the results are given in figure 5 for an undamped bellows seal (fig. 4(a)). In operation, the initial operating seal length was set at 0.525 inch (1.333 cm). Because of the shaft and housing mounting arrangement, thermal expansion tended to reduce the seal length with the result that the operating seal length varied typically from 0.525 to 0.505 inch (1.33 to 1.28 cm) during the run. These lengths were within the range specified by the manufacturer. If the bellows did not relax, the bellows force ranged between 7.5 and 12.0 pounds (33.4 and 53.4 N). Seals with and without dampers have similar spring rates since the damper exerts only a small frictional force on the bellows. In several runs a plastic diaphragm was attached to the nosepiece (fig. 4(c)); thus lubricant could be added only to the bellows and the interface would remain unlubricated, or vice versa.

Procedure

Room temperature and elevated temperature runs were made both with and without Mil-L-7808D lubricant. Seal components were cleaned ultrasonically in alcohol prior to assembly and rinsed off with alcohol after assembly. Leakage rates, temperatures, pressures, seal vibration, and speed were continuously recorded on magnetic tape.

RESULTS AND DISCUSSION

Seal Stability, Bellows Face Seals Without Mechanical Dampers

Table I lists the results obtained when operating a face seal with a bellows secondary (fig. 4(a)) without a friction damper. Inspection of the data indicate that the seal is
(1) Unstable (nosepiece vibrates) when operated without lubrication (runs 1 and 6)
(2) Stable when lubricant is present in the bellows and at the sliding interface (runs 3 and 4) before start of rotation (Lubricant flow was $1 \text{ cm}^3/\text{min}$ throughout the run.)

(3) Stable when the sliding interface (carbon nosepiece) is lubricated (no lubricant in bellows) (run 5)

(4) Stable when the bellows contains lubricant (no lubricant at interface) (run 7)

As an example of unstable operation, figure 6 shows the transducer outputs of proximity, accelerometer, and pressure probes for the first 100 milliseconds of run 1. Within 3 milliseconds after the start of rotation both the accelerometer and proximity probes indicate the onset of severe vibration. The proximity probe shows 0.010-inch (0.025-cm) displacement. This amplitude was maintained throughout the run and may have been larger if the proximity probe had not limited the nosepiece travel (also see fig. 2). In general, the frequency was high (700 Hz or higher) during startup and shutdown, but at steady speed operation of 24 000 rpm, the vibration frequency was near rotational frequency (400 Hz). This frequency-to-shaft-speed relation is illustrated in figures 7 and 8. These figures show in detail the vibration frequencies during shutdown. In run 2 (table I) rotation was started with no lubricant and starting lubricant flow after 60 seconds of operation made no detectable change in vibration amplitude or frequency at 24 000 rpm, however the peak frequency at shutdown was lower (fig. 8) than that for the run without lubricant. Except for this peak frequency difference, the two runs were identical.

The seals without mechanical dampers showed the same vibration characteristics in the elevated temperature apparatus as in the room temperature apparatus. The lubricated seals operated in a stable manner, and the unlubricated seals operated in an unstable manner. Table II lists the pertinent data for runs with nitrogen gas and with products of combustion (see APPARATUS AND PROCEDURE section). In these runs the lubricant was added before the start of rotation and was not supplied during operation. As a result, the lubricant evaporated near the end of the 600° and 800° F (589 and 700 K) temperature runs and unstable operation occurred.

Seal Stability, Bellows Face Seals With Mechanical Dampers

Bellows face seals with a mechanical damper (fig. 4(b)) were run in both the room temperature apparatus and the elevated temperature apparatus. The mechanical damper consisted of a thin steel band which rubs against the bellows outside diameter; the resulting friction serves to dampen vibration. The seals with a mechanical damper ran in a stable manner for unlubricated and lubricated operation in nitrogen gas at room temperature and in both nitrogen gas and products of combustion at elevated temperature. These data are given in table III. No unstable operation occurred with these damped seals.

Effect of Vibration on Seal Leakage

Inspection of the data in tables I and II reveal a marked increase in leakage rate when the seal operates in an unstable manner. For example (in table I) the leakage rate is less than 100 standard cubic centimeters per minute (SCCM) for stable operation and is 27 500 standard cubic centimeters per minute for unstable operation. This high leakage rate is due to the nosepiece vibration amplitude which was measured to be 0.010 inch (0.025 cm). There was evidence that the proximity probe was limiting the vibration amplitude (nosepiece insert hitting the probe). When the proximity probe had greater clearance, amplitudes of 0.030 inch (0.076 cm) were indicated.

Unstable seal operation did not cause any extremely detrimental carbon wear or damage; therefore, post-test leakage rates were low and could not be used as a criterion for determining whether the mode of prior seal operation was stable or unstable.

Seal Nosepiece Temperatures

Figure 9 shows carbon nosepiece temperature as a function of operating time. All of the runs of figure 9 were started at room temperature and no external heat was added; therefore, the increase in the nosepiece temperature is due solely to sliding contact at the seal dam. The bellows face seals with mechanical dampers operated without lubrication in a stable manner, but the nosepiece temperature exceeded 700° F (644 K) (limit of recording equipment) very quickly (160 sec in one case and 290 sec in the other) and the maximum temperature reached is unknown. However, when the seal was lubricated at 1 cubic centimeter per minute of lubricant flow, a level-off temperature of 560° F (566 K) was attained. Unstable seal operation results in lower nosepiece temperature (370° F or 461 K) because of intermittent contact. The data indicate that temperatures due to unlubricated sliding contact are high enough to reduce the hardness of the rotating seat, and reduction in seat hardness can lead to excessive wear. As an example, in an allied experiment, a 1.5-inch (2.93-cm) mean diameter seal was operated at 250 feet per second (76 m/sec) in an argon and oil mist environment. The seal seat material was 440 C stainless steel. Hardness measurements on the seal wear track after operation were 53 Rockwell C as compared to the original 57 Rockwell C. Such a reduction in hardness would require a tempering temperature of 1000° F (811 K) (ref. 5). The wear track on the seat was 0.0045 inch (0.0114 cm) deep after 30 hours of operation.

Bellows Fatigue

It is recognized that, in many applications, bellows face seals operate for thousands of hours without bellows fatigue. It is significant, therefore, when bellows fatigue is reached in only seconds of unstable operation. Figure 10 shows the time-temperature history of three seals. One which operated in an unstable mode at room temperature for 630 seconds suffered a 0.015 inch (0.038 cm) reduction (relaxation) in bellows length. The seal which operated unstably at 600° F (589 K) failed within 17.8 seconds; and the seal which operated unstably at 800° F (700 K) failed within 4.4 seconds.

SUMMARY OF RESULTS

Shaft face seals of 1.57-inch (3.99-cm) mean sliding diameter with a bellows secondary seal were run in inert and reducing gaseous environments. Measurements were made of seal stability (nosepiece vibration), leakage rate, and nosepiece temperature. The effect of viscous and coulomb damping on seal stability were studied at temperatures to 800° F (700 K) and sliding speeds to 165 feet per second (50.3 m/sec). The pertinent results are as follows:

1. Seal stability (vibration)
 - a. Seals without either friction damping or viscous damping have a very strong tendency to operate in an unstable mode at high frequencies.
 - b. Seals with either friction damping or viscous damping operate in a stable manner (no vibration).
 - c. For unstable operation the predominant frequency of vibration was near that of shaft rotational frequency (400 Hz) with excursions to near 700 hertz (or higher) during start and stop.
2. Leakage rates: Seals which operated unstably had large (0.010 in. or 0.025 cm) nosepiece vibration amplitudes; hence, leakage rates were excessive.
3. Nosepiece temperature
 - a. Unlubricated seals operating in a stable manner and at the manufacturers' recommended bellows compression generated high ($>700^{\circ}$ F or >644 K) carbon temperatures because of sliding contact at the interface; lubricated seals operating in a stable manner generated much lower (560° F or 566 K) nosepiece temperature.
 - b. Temperatures generated by sliding contact can be high enough to reduce the hardness of the seal seat material.

4. Bellows fatigue: Seals operating in an unstable manner cause bellows relaxation at low temperature (75° F or 297 K) operation and bellows cracking in very short periods of operation (4 sec) at elevated temperatures such as 800° F (700 K).

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, January 24, 1969,
120-27-04-21-22.

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TABLE I. - BELLows FACE SEAL WITHOUT MECHANICAL DAMPER; OPERATION IN ROOM TEMPERATURE APPARATUS

[Sealed gas, nitrogen; operating speed, 24 000 rpm; sliding speed at mean diameter of sliding dam, 165 ft/sec (50.3 m/sec).]

Run num- ber	Run time, sec	Lubrication	Sealed pressure		Leakage rate, SCCM ^a			Nosepiece temperature		Operation	Vibration frequency, Hz
			lbf/in. ²	N/cm ²	Pretest at 5 lbf/in. ² (3.4 N/cm ²)	Dynamic 5 lbf/in. ² (3.4 N/cm ²)	Post test at 5 lbf/in. ² (3.4 N/cm ²)	°F	K		
1	300	None	b 0.80	b 0.55	<10	27 500	12	380	466	Unstable	450
2	300	None at start; lubricant added 60 sec into run	b 0.80	b 0.55	12	26 500	10	395	475	Unstable	395
3	360	Lubricant flow started 60 sec be- fore start of run	2	1.38	10	<100	<10	524	546	Stable	---
4	540	Lubricant flow started 60 sec be- fore start of run	2	1.38	<10	<100	<10	590	583	Stable	---
5	60	No lubricant in bel- lows; carbon wetted with lubricant	2	1.38	---	----	---	---	---	Stable	---
6	30	None	2	1.38	---	----	---	---	---	Unstable	---
7	60	No lubricant on carbon nosepiece, lubricant in bellows	2	1.38	---	----	---	---	---	Stable	---

^aLeakage rate related to standard conditions of 14.7 lbf/in.² (10.1 N/cm²) and 68° F (293 K); SCCM = standard cm³/min.

^bHigh leakage due to unstable operation caused reduction in sealed pressure from 2 to 0.8 lbf/in.² (1.38 to 0.55 N/cm²).

TABLE II. - BELLows FACE SEAL WITHOUT MECHANICAL DAMPER; OPERATION IN ELEVATED TEMPERATURE APPARATUS

Run number	Run time, sec	Lubrication	Sealed pressure		Sealed gas	Nosepiece temperature		Shaft speed, rpm	Leakage rate, SCCM ^a			Operation		
			lbf/in. ²	N/cm ²		Starting	Maximum		Pretest at 5 lbf/in. ² (3.4 N/cm ²)	Dynamic	Post test at 5 lbf/in. ² (3.4 N/cm ²)			
			°F	K		°F	K		(3.4 N/cm ²)					
8	67	None	2	1.38	N ₂	80	300	340	444	5 000	<10	18 500	----	Stable 4.8 sec; unstable 62.2 sec
9	68	None	2	1.38	N ₂	140	333	332	440	5 000	----	17 200	----	Unstable
10	210 110 170	Lubricant added before start of run ^b	2	1.38	N ₂	80	300	---	---	5 000	----	650	----	Stable
						---	---	---	---	10 000	----	765	----	
						---	---	---	---	15 000	----	700	633	
11	65 79 74 84 40	Lubricant added before start of run ^b	2	1.38	N ₂	89	305	210	372	15 000	770	905	790	Stable
						200	366	368	460		790	730	870	
						400	478	416	486		870	<500	-----	
						500	533	516	542		470	<500	-----	
						600	589	659	621		220	17 200	-----	Unstable after 22 sec due to evaporation of lubricant
12	66	None	2	1.38	N ₂	200	366	429	494	15 000	<10	<500	410	Stable 64.6 sec; unstable 1.4 sec
13	4.5	None	2	1.38	N ₂	200	366	---	---	3 000	410	-----	1 600	Unstable 4 sec
14	10	None	2	1.38	POC ^d	400	478	---	---	3 000	1340	13 000	86	Stable 3.5 sec; unstable 6.5 sec
15	80 80 80 70 188	Lubricant added before start of run ^b	2	1.38	POC ^d	400	478	448	504	15 000	----	1 180	----	Stable
			4	2.76		---	---	487	526		----	1 570	----	
			6	4.14		---	---	603	590		----	1 350	----	
			8	5.58		---	---	675	630		----	1 735	----	
			2	1.38		---	---	548	560		----	<500	160	
16	80 80 60 60 64	Lubricant added before start of run ^b	2	1.38	POC ^d	600	589	665	625	15 000	<500	388	----	Stable
			4	2.76		---	---	685	636		----	429	----	
			6	4.14		---	---	754	674		----	456	----	
			8	5.52		---	---	812	706		----	741	----	
			2	1.38		---	---	799	699		----	<500	190	
17	6.6	No lubricant added	2	1.38	POC ^d	800	700	---	---	3 000	<500	-----	e ₇ 500	Stable 2.2 sec; unstable when lubricant evaporated
18	220	Lubricant added	2	1.38	POC ^d	800	700	---	---	15 000	e ₇ 130	7 690	e ₁₂ 000	Stable operation, bellows cracked during instability of run 17

^aLeakage rate related to standard conditions of 14.7 lbf/in.² (10.1 N/cm²) and 68° F (293 K); SCCM = standard cm³/min.^b2 cm³ of lubricant was added to bellows region (see fig. 3) before rotation. No further lubricant was added during run.^cAfter 22 sec.^dPOC = products of combustion: 24.7 percent CH₄, 0.7 percent CO, 0.7 percent CO₂, 40.0 percent H₂, 25.7 percent N₂, and 8.2 percent H₂O.^eCracked in bellows.

TABLE III. - RESULTS OF OPERATION OF BELLows FACE SEAL WITH MECHANICAL DAMPER

[Sealed pressure, 2 lbf/in.² (1.38 N/cm²).]

Run num- ber	Run time, sec	Lubrication	Sealed gas	Nosepiece temperature				Shaft speed, rpm	Leakage rate, SCCM ^a			Operation		
				Starting		Maximum			Pretest at 5 lbf/in. ² (3.4 N/cm ²)	Dynamic	Post test at 5 lbf/in. ² (3.4 N/cm ²)			
				°F	K	°F	K							
Room temperature apparatus														
19	300	None	N ₂	75	297	---	---	24 000	<10	<100	<10	Stable		
20	300	None	N ₂	75	297	---	---	24 000	<10	<100	<10	Stable		
21	300	None	N ₂	75	297	---	---	24 000	<10	<100	<10	Stable		
22	300	None	N ₂	75	297	>700	>644	24 000	<10	<100	<10	Stable		
23	300	None	N ₂	75	297	>700	>644	24 000	<10	<100	<10	Stable		
24	300	Lubricated	N ₂	75	297	370	461	24 000	<10	<100	<10	Stable		
25	480	Lubricated	N ₂	75	297	562	568	24 000	<10	<100	<10	Stable		
Elevated temperature apparatus														
26	66	None	N ₂	74	296	285	414	15 000	<10	<500	<10	Stable		
	67			200	366	374	463		---	530	---			
	116			400	478	555	564		---	<500	---			
	77			500	533	610	594		---	<500	---			
	69			600	589	649	616		---	500	---			
	312			600	589	797	698		---	220	---	↓		
27	68	None	POC ^d	300	422	519	544	15 000	86	<500	100	Stable		
	68			600	589	721	656		106	596	20			
	313			714	652	780	689		---	<500	95			
	280			796	698	871	739		133	<500	---	↓		

^aLeakage rate related to standard conditions of 14.7 lbf/in.² (10.1 N/cm²) and 68° F (293 K);SCCM = standard cm³/min.^bPOC = products of combustion: 24.7 percent CH₄, 0.7 percent CO, 0.7 percent CO₂, 40.0 percent H₂, 25.7 percent N₂, and 8.2 percent H₂O.

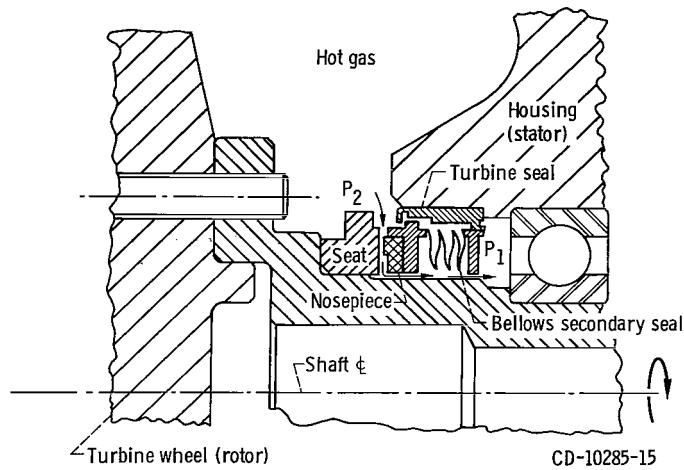


Figure 1. - Cross-sectional view of face seal in turbopump.

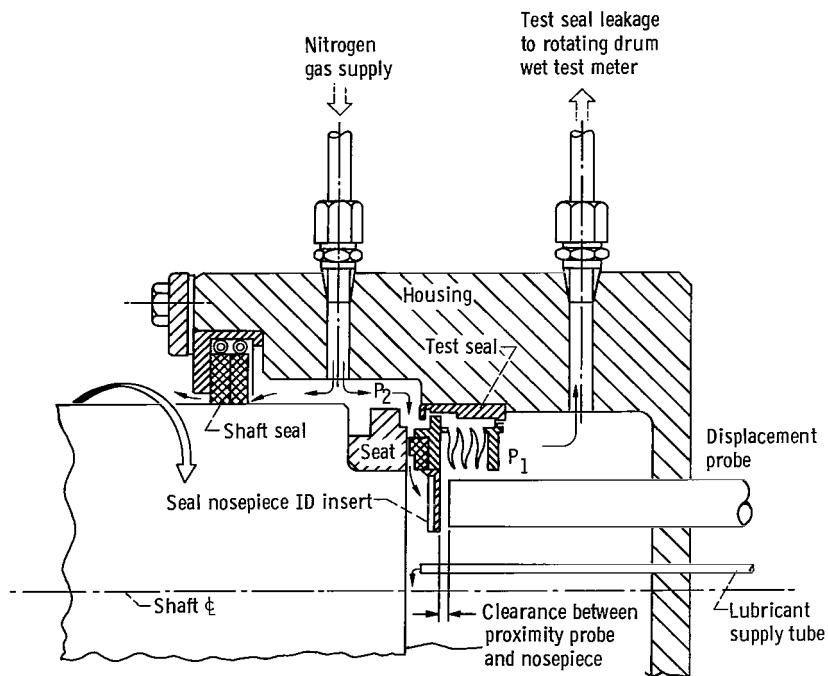


Figure 2. - Cross-sectional view of room temperature seal test apparatus.

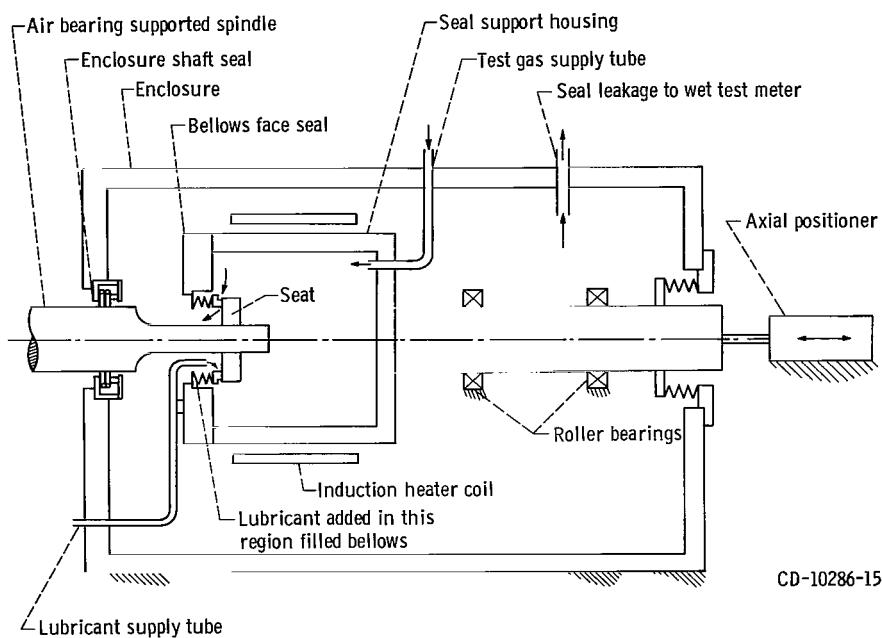
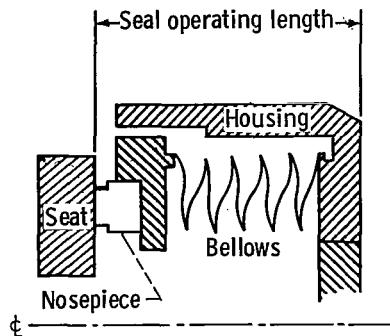
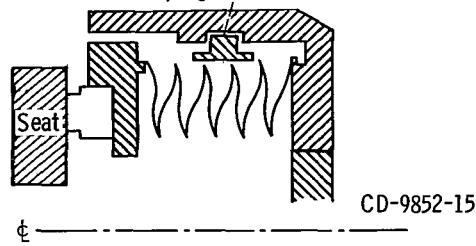


Figure 3. - Elevated temperature seal test apparatus.

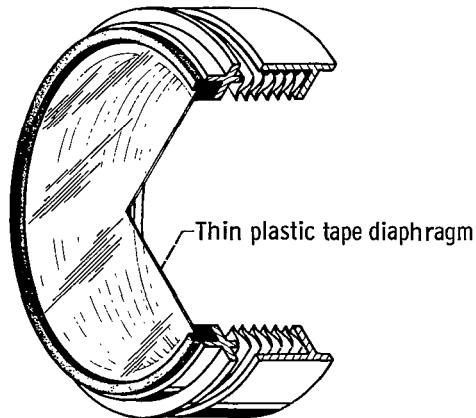


(a) Face seal with bellows secondary seal.

Circumferential mechanical
friction damper (Coulomb damping)



(b) Face seal with damper on bellows.



(c) Bellows face seal with diaphragm to separate interface area from bellows area.

Figure 4. - Shaft face seals.

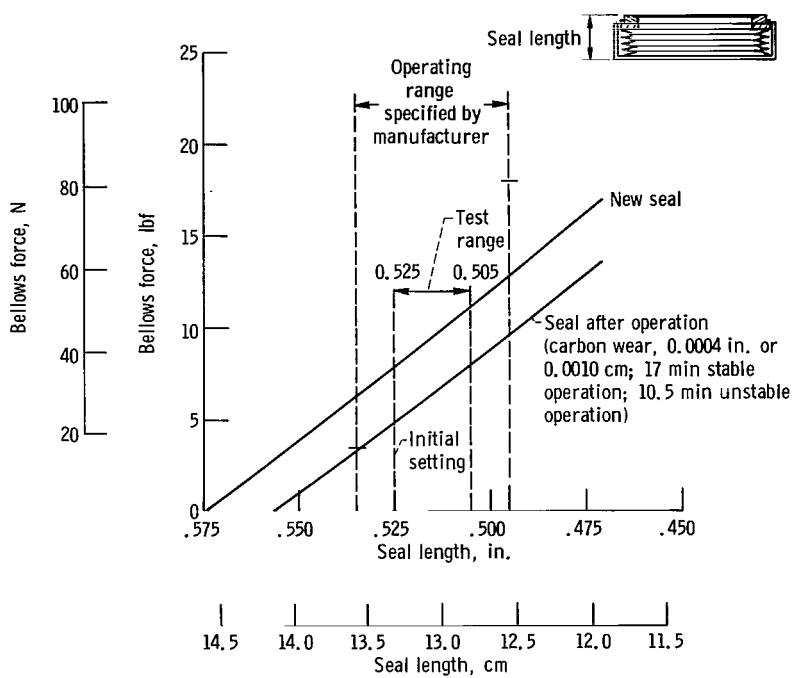


Figure 5. - Bellows spring rate,

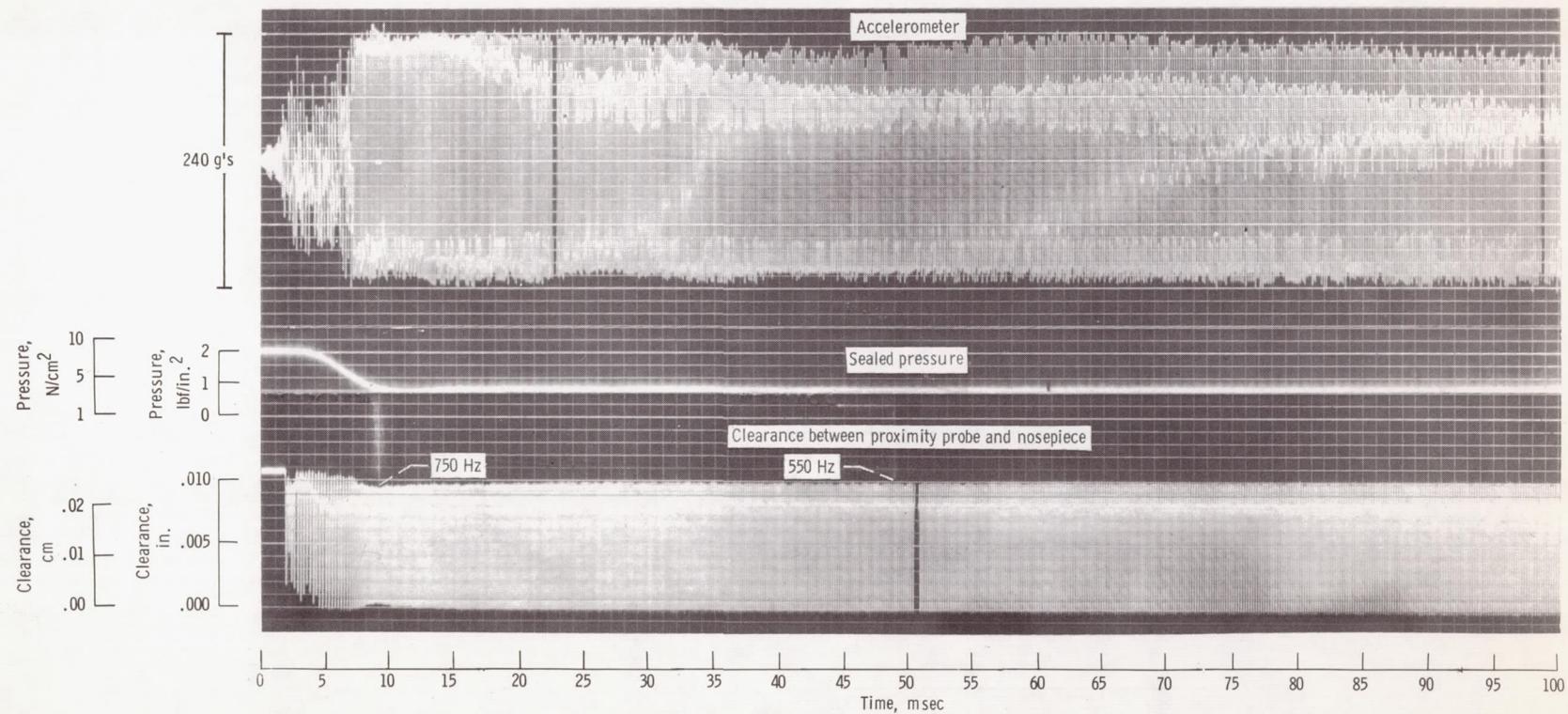


Figure 6. - Typical transducer outputs of accelerometer, pressure, and proximity probe for a seal during unstable operation. No lubricant; sealed gas, nitrogen at room temperature.

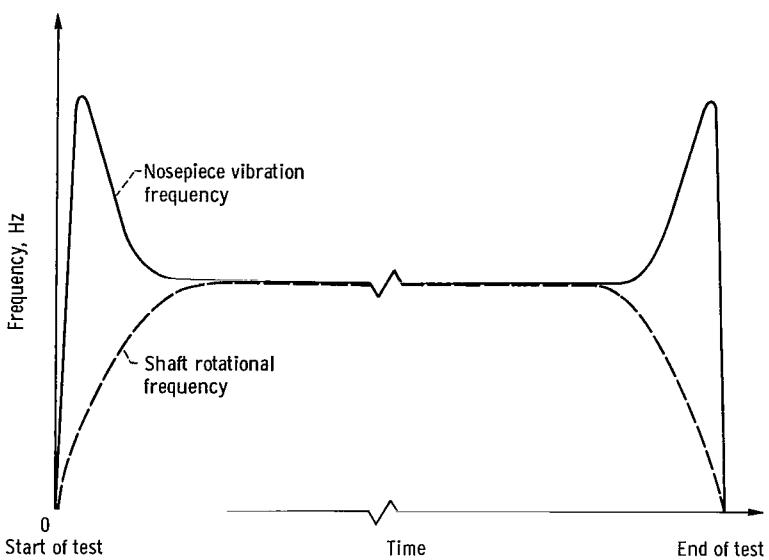


Figure 7. - Typical nosepiece and shaft rotational frequency relation exhibited during unstable seal operation. Sealed gas, nitrogen; no external heat added; no lubricant.

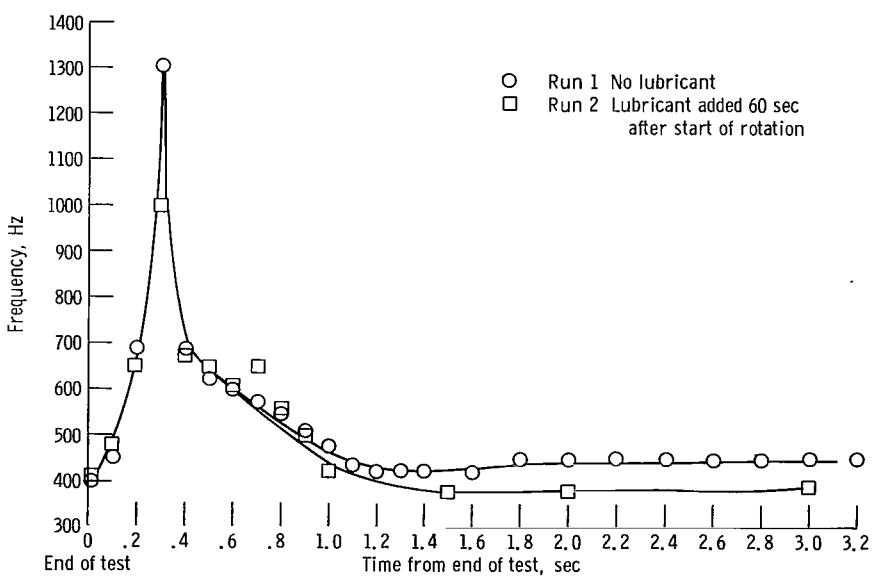


Figure 8. - Predominate seal vibrational frequency during shutdown.

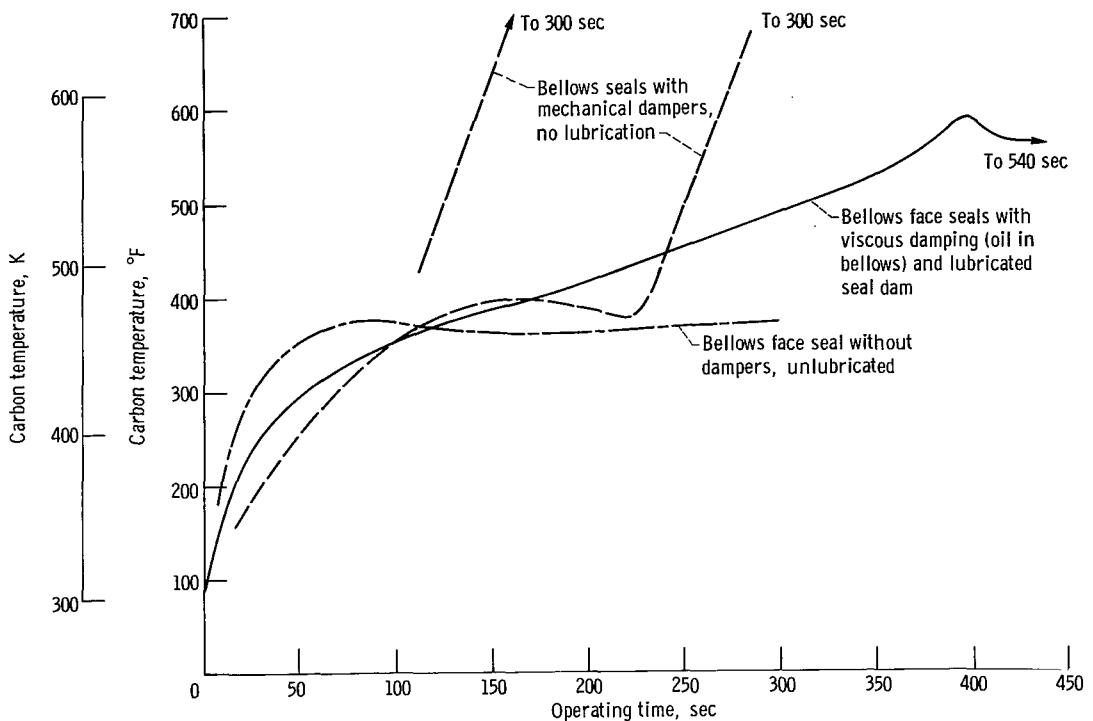


Figure 9. - Carbon nosepiece temperatures. Sealed gas, nitrogen; shaft speed, 24 000 rpm; no external heat.

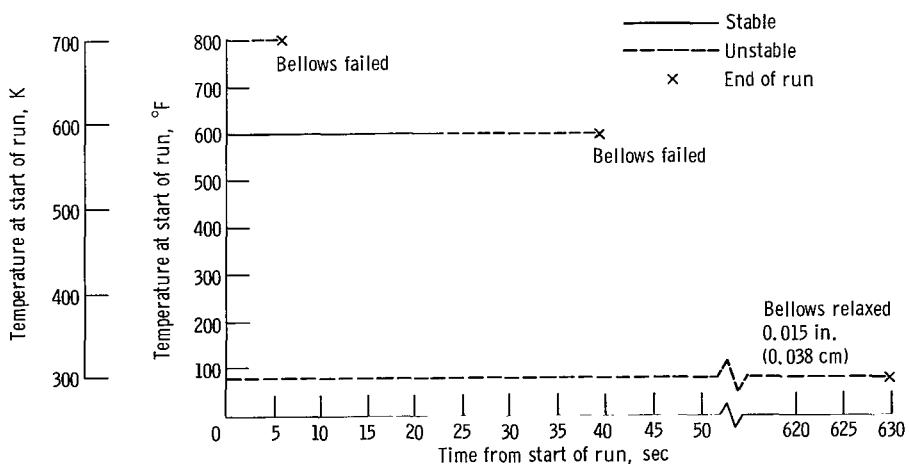


Figure 10. - Bellows fatigue as function of time and temperature. No lubrication; no mechanical or viscous damping.

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